Chemical and Petroleum Engineering, Vol. 35, Nos. 1–2, 1999

OIL AND GAS EQUIPMENT

EFFICIENCY OF THE HEAT-RECOVERY STEAM TURBINE CIRCUIT IN THE COMPRESSOR INSTALLATIONS OF GAS TRANSPORT SYSTEMS

V. I. Evenko

UDC 536.27.001.5

Compressor installations with gas-turbine drive have become common in systems for transport of natural and petroleum gas. However, the thermal economy of the gas-turbine motors used is low. One reason for this is that 60% of the heat of combustion of the fuel gas used by the motor is lost with the motor discharge gases. In addition, the heat removed from the compressed gas in air-cooling apparatuses (ACA) is also dissipated in the atmosphere.

Recovery of heat losses for the purpose of partial conversion of the heat lost to work would allow significantly increasing the efficiency of utilization of the heat of combustion of fuel gas.

This recovery can be executed in a steam-turbine circuit. The skeleton diagram and cycle of a gas-turbine installation (GTI) with a heat-recovery steam-turbine circuit are shown in Fig. 1.

Consider the basic premises of the method of calculating the efficiency of heat recovery in a steam-turbine circuit.

Let us define the values of the parameters of the steam – pressure p_{1s} and temperature t_{1s} after the high-pressure boiler-recovery unit (HPBR) and the temperature difference $\Delta t = t_o - t_{4s}$ (see Fig. 1). Then the amount of steam g_{s1} processed by the HPBR on conversion to 1 kg of discharged gases will be

$$g_{s1} = \frac{c_{pt_{(4\text{-o})}}(t_4 - t_o)}{h_{1s} - h_{4s}},$$

where $t_o = t_{4s} + \Delta t$; t_{4s} is the boiling point of water at pressure p_{1s}; $c_{pt_{(4\text{-o})}}$ is the average isobaric heat capacity of the discharge gases in the temperature range from t_4 to t_o; h_{1s}, h_{4s} are the enthalpy of the steam and boiling water, respectively (at pressure p_{1s}).

The efficiency of the high-pressure boiler recovery unit is

$$G_{s1} = Gg_{s1},$$

where G is the amount of discharge gases.

The temperature of the gas at the outlet from the HPBR is

$$t_5 = t_o - \frac{g_{s1}(h_{4s} - h_{3s})}{c_{pt_{(o\text{-}5)}}},$$

where h_{3s} is the enthalpy of the condensate entering the HPBR (boiling water at pressure p_{2s} in the condenser).

The work of 1 kg of steam in the turbine is

$$l_{s1} = (h_{1s} - h_{2s})\eta_t,$$

Bryansk State Technical University. Translated from Khimicheskoe i Neftegazovoe Mashinostroenie, No. 1, pp. 19–22, January, 1999.

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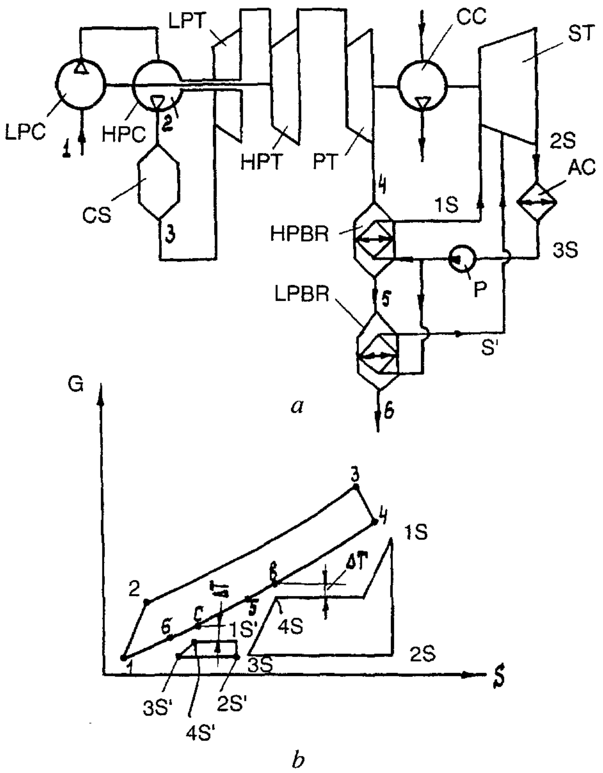

Fig. 1. Diagram (a) and cycle (b) of a gas-turbine installation with a heat-recovery steam power circuit: LPC) low-pressure compressor; HPC) high-pressure compressor; LPT) low-pressure turbine; HPT) high-pressure turbine; PT) power turbine; CC) centrifugal compressor; HPBR) high-pressure boiler recovery unit; ST) steam turbine; AC) air condenser; P) pump.

where h_{2s} is the enthalpy of the steam after isentropic expansion in the turbine to pressure p_{2s}; η_t is the adiabatic efficiency of the steam turbine.

The discharge gases from the HPBR with temperature t_5 enter the low-pressure boiler recovery unit (LPBR). We will assume that the LPBR produces dry steam with a pressure of $p_{1s} = 0.1$ MPa. Then the amount of steam developed by LPBR on conversion to 1 kg of discharge gases will be

$$g_{s2} = \frac{c_{pt_{(5\text{-}o)}}(t_5 - t_o)}{h_{1s'} - h_{4s'}},$$

where $t_d = t_{4s} + \Delta t$.

The efficiency of LPBR is

$$G_{s2} = G g_{s2}.$$

The temperature of the discharge gases at the outlet from LPBR is

$$t_6 = t_d - \frac{g_{s2}(h_{4s} - h_{3s})}{c_{pt_{(d\text{-}6)}}}.$$

The work executed by 1 k of steam in the turbine is

TABLE 1. Results of Calculating the Efficiency of Recovery of Heat from Discharge Gases of the NK-16ST Electric Motor

p_{1s}, Pa	$1{\cdot}10^6$	$1.5{\cdot}10^6$	$2{\cdot}10^6$	$2.5{\cdot}10^6$
g_{s1}, kg steam/kg gas	0.083	0.078	0.0745	0.0704
G_{s1}, kg/sec	8.55	8.03	7.67	7.256
t_5, °C	143	159	171	181
l_{s1}, kJ/kg steam	611.2	672	694.4	710.4
g_{s2}, kg steam/kg gas	0.0135	0.02	0.025	0.029
G_{s2}, kg/sec	1.39	2.06	2.58	2.99
t_6, °C	107	105	104	103
l_{s2}, kJ/kg steam	272	272	272	272
q_{a1}, kJ/kg gas	186.7	169.6	159.3	148.3
q_{a2}, kJ/kg gas	29.9	44.2	55.3	64.1
l_{at1}, kJ/kg gas	7.99	7.26	6.82	6.35
l_{at2}, kJ/kg gas	1.28	1.89	2.37	2.74
N_{s1}, kW	4150	4404	4376	4257
N_{s2}, kW	224	333	441	483
$N_s = N_{s1} + N_{s2}$, kW	4374	4737	4817	4740
η_Σ	0.3654	0.37	0.3695	0.3649
η_Σ^{1k}	0.3694	0.376	0.3775	0.3736
$\eta_\Sigma^{1k}/\eta_{e\,\mathrm{GTI}}$	1.26	1.276	1.274	1.258
$\eta_\Sigma/\eta_{e\,\mathrm{GTI}}$	1.274	1.297	1.302	1.288

$$l_{s2} = (h_{1s} - h_{2s})\eta_t.$$

The amount of heat q_a removed from the condensed steam in the condenser on conversion to 1 kg of discharged gases is

$$q_a = q_{a1} + q_{a2} = g_{s1}(h_{2d} - h_{3s}) + g_{s2}(h_{2d} - h_{3s})$$

or

$$q_a = q_{a1} + q_a = g_{s1}(h_{1s} - l_{s1} - h_{3s}) + g_{s2}(h_{1s} - l_{s2} - h_{3s});$$

where h_{2d} is the enthalpy of the steam at the outlet of the turbine (with consideration of the losses from the irreversibility of the actual expansion process).

The energy P_f consumed by the air condenser fans at the ambient average annual temperature is determined by the equation

$$l_{a.t} = \frac{g_{a.t}\Delta p_t}{\rho_{a.t}\eta_f \cdot 1000},$$

where $g_{a.t}$, Δp_t is the amount of air passing through the condenser and the air drag of the condenser system (averaged over a year); $\rho_{a.t}$ is the density of the atmospheric air at the average annual temperature; η_f is the efficiency of the fan.

Then

$$g_{a.t} = \frac{q_a}{C_{pt}\delta t_{a.t}},$$

where $\delta t_{a.t}$ is the increase in the air temperature in the condenser (at the average annual temperature of the atmospheric air).

TABLE 2. Results of Calculating the Efficiency of Recovery of Heat Removed from Gas in the Cooling System of the TKA-Ts-16/120 Installation

TKA compressor installation	Without recovery	With steam-turbine circuit
Amount of heat removed into the atmosphere, kJ/kg gas		
in ACA1	291	154.7
in ACA2	359	175.7
in the condenser	–	286.5
Amount of air passing through the ACA and condenser, kg/sec		
ACA1	15.3	8.1
ACA2	18.9	9.2
condenser	–	15.1
Energy used by fans, l_f, kJ/kg gas		
ACA1	12.4	6.6
ACA2	15.4	7.5
condenser	–	12.3
Consumption of electrical power for driving fans E_f, kW·h/year		
ACA1	$2.93 \cdot 10^6$	$1.56 \cdot 10^6$
ACA2	$3.64 \cdot 10^6$	$1.77 \cdot 10^6$
condenser	–	$2.91 \cdot 10^6$
Amount of steam generated G_s, kg/sec	–	3.3
Work of steam turbine l_t, kJ/kg gas	–	32.9
Amount of electrical power produced E_p, kW·h/year	–	$7.02 \cdot 10^6$
E_s - E_f, kW·h/year	$-6.57 \cdot 10^6$	$0.77 \cdot 10^6$

The effective power of the steam-turbine circuit is

$$N_{es} = G[(g_{s1} l_{s1} + g_{s2} l_{s2})\eta_m - l_a/\eta_{em}],$$

where η_m is the mechanical efficiency of the turbine; η_{em} is the efficiency of the electric motor of the fan.

The efficiency of the GTI with recovery of the heat from the discharge gases is

$$\eta_\Sigma = \frac{N_e + N_{s1}}{G_t Q_n^p},$$

where N_e is the effective power of the GTI; G_t is the consumption of fuel gas; Q_n^p is the head of combustion of the gas.

If the steam-turbine circuit is only set up with HPBR (there is no LPBR), the effective efficiency of the GTI with recovery will be

$$\eta_\Sigma^{1\kappa} = \frac{N_e + N_{s1}}{G_t Q_n^p},$$

where $N_{s1} = G(g_{s1} l_{s1} \eta_m - l_{a.t}/\eta_{em})$.

The efficiency of recovery of heat from the discharge gases of a NK-16ST gas-turbine engine (effective power N_3 = 16,000 kW; cycle air consumption of G_c = 102 kg/sec; fuel gas consumption of G_t = 1.103 kg/sec; gas combustion heat $Q_n{}^p$ = 50,000 kJ/kg; discharge gas temperature of 645 K; efficiency of GTI of η_{eGTI} = 0.29) was calculated for the following data: $\Delta t = t_o - t_{4s} = t_a - t_{4s}$ = 10°C; t_{1s} = 300°C; $p_{2s} = p_{2s}$ = 0.01 MPa; the adiabatic efficiency of the steam turbine is η_t = 0.8;

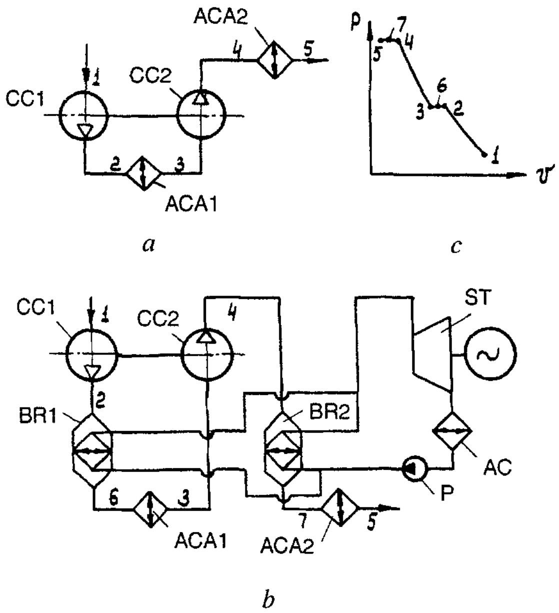

Fig. 2. Diagram of a petroleum gas compressor station cooling system: *a*) without recovery; *b*) with a steam-turbine recovery circuit; *c*) illustration of the working process of the compressor installation; CC1, CC2) low- and high-pressure centrifugal compressors; ACA1, ACA2) air-cooling apparatuses; BR1, BR2) boiler recovery units; ST) steam turbine; AC) air condenser; P) pump.

the mechanical efficiency of the steam turbine is $\eta_m = 0.96$; the efficiency of the electric motor is $\eta_{em} = 0.95$; the average annual ambient temperature is $t_{am} = 5°C$; $\delta t_t = 19°C$; $\Delta p_t = 610$ Pa; the amount of discharge gases is $G = G_i + G_t = 103$ kg/sec. The properties of the water and water vapor were determined with the tables in [1].

The results reported show that recovery of heat from GTI discharge gases in a steam-turbine circuit significantly increases the efficiency of the installation. The vapor pressure p_{1s} has an optimum with respect to the efficiency of recovery. In the case examined, $p_{1s.opt} = 2$ MPa. Introduction of LPBR as a second stage of cooling of the discharge gases causes a further increase in the efficiency of the installation.

Two variants of utilizing the power of a steam turbine are possible: the first one for gas transport, the second for generation of electrical power.

In the first variant, the steam turbine is connected with the shaft of a centrifugal compressor (see Fig. 1). The powers of the gas N_{GTI} and steam N_{ST} turbines are respectively

$$N_{GTI} = \frac{N_c}{1 + N_s / N_c} = 1298\,\text{kW}; \quad N_{st} = N_{GTI}\left(\frac{N_s}{N_c}\right) = 3702\,\text{kW},$$

and the consumption of fuel gas will be reduced to

$$G_{TI} = N_{GTI}/(\eta_{GTI} Q_n^{\,p}) = 0.848\ \text{kg/sec}$$

(the decrease in fuel consumption causes a decrease in consumption of cycle air and the pressure ratio π_k so that η_{GTI} will decrease slightly. This situation will not be considered further and η_{GTI} will remain unchanged with a small error.

The annual consumption of fuel gas by one gas pumping unit with a NK-16ST motor will be decreased by $\Delta V = (G_t - G_{ti})\, 3600 \tau v_0 = 11.26 \cdot 10^6$ m^3/year (here τ is the number of hours in a year, $\tau = 8760$ h; $v_0 = 1.4$ m^3/kg is the specific volume of gas in normal conditions). If there is no LPBR in the steam-turbine circuit, then

$$N_{GTI} = \frac{N_c}{1 + N_{s1} / N_c} = 12\,564\,\text{kW};$$

$$N_{st} = N'_{GTI}\left(\frac{N_{s1}}{N_c}\right) = 3436\,\text{kW},$$

$$G'_{TI} = N'_{GTI}/\eta_{GTI} Q_n^{\,p} = 0.8665 \text{ kg/sec},$$

$$\Delta V' = (G_t - G'_{TI}) 3600 \cdot \tau v_0 = 10.44 \cdot 10^6 \text{ m}^3\text{/year}.$$

Introduction of a LPBR thus decreases consumption of fuel gas by $\Delta V_{LPBR} = \Delta V - \Delta V' = 0.82 \cdot 10^6$ m^3/year.

In the second variant, the steam turbine is connected to the shaft of an electric generator. The amount of electrical power that will be generated in a year will be $E = N_s \eta_{eg} \tau = 40 \cdot 10^6$ kW·h/year (here η_{eg} is the efficiency of the electric generator).

Due to recovery of the heat of discharge gases, a compressor station can be converted from consumption of electrical power to generation of electrical power.

In the transport of natural gas, its temperature after the compressor is below 100°C, so that it is currently almost impossible to produce mechanical energy due to recovery of heat removed from gas in the ACA of a natural gas compressor station.

However, the temperature of petroleum gas at the outlet from the first and second units of the compressor installation attains 160–170°C, which allows recovering the heat removed from the gas in the cooling system with a steam-turbine circuit [2].

A diagram of a petroleum gas cooling system with a recovery circuit is shown and the working process of a two-stage compressor is illustrated in Fig. 2.

Without recovery in the cooling system, heat $q_{2\text{-}3} = c_{pg}(t_2 - t_3)$ and $q_{4\text{-}5} = c_{pg}(t_4 - t_5)$, where c_{pg} is the average heat capacity of the gas, will be removed from 1 kg of gas in ACA1 and ACA2 in the cooling system.

When a steam-turbine circuit is introduced, heat is removed from the gas in the boiler recovery units (BR)

$$q_{2\text{-}6} = c_{pg}(t_2 - t_6); \quad q_{4\text{-}7} = c_{pg}\,(t_4 - t_7),$$

and air-cooling apparatuses

$$q_{6\text{-}3} = c_{pg}(t_6 - t_3); \quad q_{7\text{-}5} = c_{pg}\,(t_7 - t_5).$$

The efficiency of BR is

$$g_{s1} = \frac{q_{2\text{-}6}}{h_{1s} - h_{3s}}; \quad g_{s2} = \frac{q_{4\text{-}7}}{h_{1s} - h_{3s}}$$

or $G_s = G_g\,(g_{s1} + g_{s2})$, where G_s is the amount of steam generated by the boiler recovery units; G_g is the amount of petroleum gas passing through the cooling system.

The amount of heat removed in the air condenser (on conversion to 1 kg of gas) is $q_a = q_{2\text{-}6} + q_{4\text{-}7} - l_T$, where $l_t = (g_{s1} + g_{s2})\,(h_{1s} - h_{2s})\eta_T$ is the work of the steam turbine per 1 kg of gas.

The amount of electrical power generated by the recovery circuit is

$$E_p = G_g l_t / \eta_{em} \tau.$$

The efficiency of recovery of the heat removed from petroleum gas in the cooling system of a TKA-Ts-61/120-(I+II) compressor installation with an output of $G_g = 25.6$ kg/sec and required power of $N_k = 16{,}000$ kW was calculated with the gas parameters: $p_1 = 0.59$ MPa; $T_1 = 293$ K; $p_2 = 3.23$ MPa; $T_2 = 432$ K; $p_3 = 3.23$ MPa; $T_3 = 313$ K; $p_4 = 12.77$ MPa; $T_4 = 422$ K; $p_5 = 12.77$ MPa; $T_5 = 313$ K. The following were used in the calculation: $p_{1s} = 0.1$ MPa; $t_{1s} = 100$°C (dry steam); $p_{2s} = 0.01$ MPa.

As the results suggest, recovery of heat removed from petroleum gas in the cooling system of one TCA-Ts-16/120 aggregate allows delivering $0.77 \cdot 10^6$ kW·h/year to the network instead of removing $6.57 \cdot 10^6$ kW·h/year from the electric network. This means that the recovery examined saves electric power in the amount of $\Delta E = 7.34 \cdot 10^6$kW·h/year.

With consideration of recovery of the heat from discharge gases in the gas-turbine drive, the total savings of electric power are $\Delta E_\Sigma = 7.3 \cdot 10^6$ kW·h/year per TKA-Ts-16/120 aggregate.

Another advantage of recovery of heat with the steam-turbine circuit is the possibility of operation of the gas-turbine drive at full power with the steam-turbine circuit switched off, which allows repairing the steam-turbine circuit equipment without shutting down the gas-pumping aggregate.

REFERENCES

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2. RF Patent No. 2065089, *Compressed Gas Cooling System in a Compressor Installation.*